RM E54F23

RESEARCH MEMORANDUM

INVESTIGATION OF TRANSONIC TURBINE DESIGNED FOR ZERO DIFFUSION OF SUCTION-SURFACE VELOCITY

By Warren J. Whitney, Daniel E. Monroe, and Robert Y. Wong

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

WASHINGTON

August 24, 1954
Declassified June 24, 1958

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

RESEARCH MEMORANDUM

INVESTIGATION OF TRANSONIC TURBINE DESIGNED FOR ZERO DIFFUSION OF SUCTION-SURFACE VELOCITY

By Warren J. Whitney, Daniel E. Monroe, and Robert Y. Wong

SUMMARY

The experimental performance of a transonic turbine designed for zero diffusion was determined. The results obtained with this turbine are compared with the results that were obtained in previous investigations of two other transonic turbines having diffusion parameters of 0.15 and 0.30.

The total-pressure-ratio efficiency of the transonic turbine designed for zero diffusion was between 0.86 and 0.87 at design work output and design speed, and the highest efficiency was 0.87. The rating efficiency for this turbine at design work output and design speed was between 0.85 and 0.86. These values represent an efficiency increase of about $1\frac{1}{2}$ points over that of the 0.15-diffusion turbine and $5\frac{1}{2}$ points over that of the 0.30-diffusion turbine. Although this efficiency increase was not uniquely caused by the decrease in the diffusion parameter since other design variables were changed among the three turbine designs, it was indicated that the diffusion parameter is an important design consideration that affects the over-all turbine performance. The design-point performance of the zero-diffusion turbine and the other two transonic turbines were also compared on a specific-blade-loss basis in order to eliminate the effect of varying solidity among the three turbines. The comparison indicated that the specific blade loss changed slightly between diffusion parameters of zero and 0.15 and changed substantially between diffusion parameters of 0.15 and 0.30.

INTRODUCTION

The characteristics of high specific work and high mass flow per unit area have made the transonic turbine appear attractive as a jet-engine component. The NACA Lewis laboratory has therefore been engaged in a research program that is directed toward obtaining efficiencies with the transonic turbine (a turbine designed to operate with a rotor-hub-inlet relative Mach number of approximately unity) that are comparable

with the efficiencies of contemporary jet-engine turbines which are of more conservative design. The performance of the first transonic turbine investigated in this program is reported in reference 1. This configuration had a total-pressure-ratio efficiency η_t of 0.85 at the design point and was designed for a suction-surface diffusion parameter D of 0.15, where D is defined as

$$\left(\frac{\text{maximum blade surface relative velocity - blade outlet relative velocity}}{\text{maximum blade surface relative velocity}}\right).$$

A second transonic turbine, which was designed for a suction-surface diffusion parameter of 0.30, was investigated (ref. 2), and the total-pressure-ratio efficiency η_t of this configuration was 0.81 at the design point. From the results of the investigations of references 1 and 2, it was indicated that high diffusion might be associated with increased turbine losses that impair the over-all turbine performance. Recent research (e.g., ref. 3) has indicated that compressor blade losses can also be correlated with a similar diffusion factor. For a typical compressor-blade-loss correlation curve the blade loss increases gradually until the critical diffusion value is attained and then the loss increases rapidly with increasing diffusion. From the results of the investigations of references 1 and 2, it would appear that the over-all turbine performance could be improved by employing a low rate of diffusion; the degree of improvement, however, is not known. Another transonic turbine was therefore designed to operate with a very low (substantially zero) diffusion parameter, and its experimental performance was determined. In addition to the over-all performance, detailed radial and circumferential surveys were made downstream of the rotor at design operating conditions.

The purpose of this report is to present the results of the investigation of the transonic turbine designed for zero diffusion and, from a comparison of these results with those of the two reference investigations, to determine the effect of zero diffusion on over-all turbine performance.

SYMBOLS

The following symbols are used in this report:

D diffusion parameter defined as

$$\left(\frac{\text{maximum blade surface relative velocity - blade outlet relative velocity}}{\text{maximum blade surface relative velocity}}\right)$$

$\Delta h'$ specific work output, Btu/lb

L specific blade loss defined as $\left(\frac{1 - \eta_t}{\sigma_m}\right)$

N rotative speed, rpm

p absolute pressure, lb/sq ft

p_x' total pressure defined as sum of static pressure plus pressure corresponding to axial component of absolute gas velocity

r radius, ft

U blade velocity, ft/sec

V absolute gas velocity, ft/sec

W relative gas velocity, ft/sec

w weight flow, lb/sec

γ ratio of specific heats

δ ratio of inlet-air total pressure to NACA standard sea-level pressure p_0'/p^*

ε function of γ, γ^*/γ $\left[\dfrac{\left(\dfrac{\gamma + 1}{2}\right)^{\frac{\gamma}{\gamma-1}}}{\left(\dfrac{\gamma^* + 1}{2}\right)^{\frac{\gamma^*}{\gamma^*-1}}}\right]$

η_l local adiabatic efficiency based on total-state measurements from surveys downstream of rotor

η_t total-pressure-ratio adiabatic efficiency defined as ratio of turbine work based on torque, weight flow, and speed measurements to ideal work based on inlet total temperature, inlet total pressure, and outlet total pressure

η_x rating adiabatic efficiency defined same as η_t except turbine-outlet total pressure, which is defined as sum of static pressure plus pressure corresponding to axial component of absolute gas velocity

θ_{cr} squared ratio of critical velocity at turbine inlet to critical velocity at NACA standard sea-level temperature, $(V_{cr,0}/V_{cr}^*)^2$

σ solidity, ratio of blade chord (see table I) to blade pitch

Subscripts:

0 station upstream of stator (all stations shown in fig. 1)

1 station at throat of stator passage

2 station at outlet of stator just upstream from trailing edge

3 station at free-stream condition between stator and rotor

4 station at throat of rotor passage

5 station at outlet of rotor just upstream from trailing edge

6 station downstream from turbine

cr conditions at Mach number of unity

m mean radius

t tip radius

x axial direction

Superscripts:

* NACA standard conditions

' total state

TURBINE DESIGN

Design requirements. - The design requirements for the 14-inch cold-air turbine are the same as those given in reference 1 and are included herein for convenience:

Equivalent specific work, $\Delta h'/\theta_{cr}$, Btu/lb	22.6
Equivalent weight flow, $\varepsilon w\sqrt{\theta_{cr}}/\delta$, lb/sec	11.95
Equivalent tip speed, $U_t/\sqrt{\theta_{cr}}$, ft/sec	597

Design procedure. - The turbine had the same design velocity diagram (fig. 1) and utilized the same stator as the first transonic turbine investigated (ref. 1). The assumptions used in obtaining the design velocity diagram are included herein for convenience:

1. Free vortex flow out of stator and downstream of rotor

2. Simple radial equilibrium throughout rotor and out of stator

3. A 3-percent total-pressure loss across the stator

4. An over-all efficiency of 0.88 based on total-pressure ratio to obtain the turbine-outlet total state and velocity diagrams at station 6

The design procedure was the same as that described in reference 1 with the following exception: The mass flow was integrated across the rotor-blade passages at the various axial stations by using the suction- and pressure-surface velocities and assuming a linear circumferential static-pressure variation between the two surfaces instead of assuming a linear velocity variation. The difference in blade shape affected by the difference in the two assumptions, however, is felt to be small. The hub section was designed to have the midchannel velocity decrease to a W/W_{cr} value of approximately 0.75 (arbitrarily selected) near the mid-axial position and to maintain the suction-surface velocity nearly constant at a W/W_{cr} value of 1.0 (fig. 2(a)). It was found that 37 blades were required to obtain this velocity distribution. For the first trial, the midchannel velocity distributions at the mean and tip sections were estimated by using the aforementioned hub midchannel velocity distribution and the hub, mean, and tip midchannel velocity distributions from the turbine of reference 1 as follows: At any particular axial station the differences in midchannel velocities (between hub and mean, and between hub and tip) that were calculated in the design of the reference turbine were applied to the hub midchannel velocity of the subject turbine. With the use of the estimated midchannel velocity variations at the mean and tip sections, the suction and pressure surfaces were laid out to obtain a suction-surface velocity that increased to approximately the blade-outlet velocity and remained nearly constant thereafter, as shown in figure 2.

After the mean and tip sections were evolved as a first approximation, the weight flow was integrated at the various axial stations, and the mean- and tip-section midchannel velocities were determined. The mean- and tip-section midchannel velocities were close to the estimated values; however, minor alterations in the blade shapes were required to hold the integrated weight flow within ±1 percent of design value. The resulting solidities of the hub, mean, and tip sections were 3.43, 2.86, and 2.55, respectively. The final design velocity distributions are shown in figure 2, and the rotor-blade coordinates are listed in table I. It can be noted on figure 2 that the maximum surface critical velocity ratio W/W_{cr} for this design is about 1.08, whereas for the two reference turbines the maximum ratio was 1.25. The maximum critical velocity ratio of 1.08 results from the design specification of zero diffusion, which limits the maximum surface velocity to the blade outlet value (station 5). A sketch showing the stator- and rotor-blade profiles and flow channels is shown in figure 3, where the high solidities can be noted.

APPARATUS, INSTRUMENTATION, AND METHODS

The apparatus, instrumentation, and methods of calculating the performance parameters used herein are the same as those described in reference 1. A schematic diagram of the test facility is shown in figure 4, and a photograph of the turbine and shaft assembly is shown in figure 5. Test runs were made from 30 to 130 percent design speed in even increments of 10 percent. For each speed the rating total-pressure ratio $p_0'/p_{6,x}'$ was varied from approximately 1.4 to the limiting-loading value. Inlet conditions were maintained constant at nominal values of 145° F and 32 inches of mercury absolute.

The precision of the measured quantities is estimated as follows:

Temperature, °F	±0.5
Pressure, in. Hg	±0.05
Speed, rpm	±10
Torque, percent design	±0.5

RESULTS

Over-all performance. - The over-all performance of the transonic turbine designed for a zero-diffusion parameter is presented in figure 6(a) with equivalent specific work $\Delta h'/\theta_{cr}$ shown as a function of the weight flow - speed parameter $\varepsilon wN/\delta$ for the various speeds with contours of total-pressure-ratio efficiency η_t and total-pressure ratio p_0'/p_6' superimposed. The efficiency at design work and design speed was between 0.86 and 0.87 and the maximum efficiency was 0.87. These values represent an improvement from 1 to $1\frac{1}{2}$ efficiency points over the performance of the first transonic turbine (ref. 1). The performance map of the turbine based on rating total-pressure ratio $p_0'/p_{6,x}'$ is shown in figure 6(b). This figure is included because jet-engine turbines are usually rated in this manner. As can be seen in figure 6(b), the efficiency at design work output was between 0.85 and 0.86, and the maximum efficiency was somewhat higher than 0.86. The over-all performance of the zero-diffusion turbine represents an improvement of about $1\frac{1}{2}$ efficiency points over that of the first transonic turbine (D = 0.15) and about $5\frac{1}{2}$ points over that of the second transonic turbine (D = 0.30). This effect is not entirely the result of the change in design diffusion parameter, however, as will be explained in the section DISCUSSION.

Survey results. - Detailed radial and circumferential surveys of total temperature and total pressure were made downstream of the turbine

rotor with the turbine set at design speed and design work. The results of these surveys are shown in figure 7 as contours of local adiabatic efficiency η_l. Although the absolute value of the efficiencies may be somewhat in question because the measurements were obtained in a highly unsteady-flow field, the trends exhibited by these efficiencies are significant. The efficiency contour plot appears similar to those obtained for the turbines of references 1 and 2 in that low-efficiency regions can be observed that result from loss regions in the stator, as shown in reference 4, and a general drop-off in efficiency occurred in the tip region. The efficiencies are affected not only by losses occurring in the rotor but also by losses occurring in the stator that carry through the rotor.

In an attempt to partially eliminate the effect of the stator losses, the maximum local adiabatic efficiencies at the various radii were plotted against radius ratio for the three turbines in figure 8. The general decrease in efficiency level for the outer portion of the blade height was not as great for the subject turbine as for the two reference turbines. This effect is felt to result to some extent from the lower diffusion employed. As discussed in reference 2, it was believed that centrifuging of the high-loss, low-velocity fluids toward the tip region caused an increase in the measured losses in this region. Furthermore, if the radial transfer of the loss fluid occurs well inside the rotor passage, the interference effect of these loss fluids on the main flow could cause some deterioration of the performance of the blade section at the outer radii. For the subject turbine it is felt, because of the lower diffusion and lower velocity level, that the flow conditions in the blade passage were less conducive to boundary-layer growth and separation. It is also believed that if high-loss fluids were centrifuged toward the tip section in the rotor-blade passage, the interference effect would be less serious for the subject turbine because of its lower diffusion.

DISCUSSION

In order to attain zero diffusion for the subject turbine with the prescribed midchannel velocity distributions, it was necessary to use 37 blades in the rotor design. This resulted in a mean-radius solidity of 2.86 as compared with 2.81 for the turbine of reference 1 and 2.16 for the turbine of reference 2. Since the rotor-blade flow-channel viscous loss would be expected to vary as the ratio of wetted surface area to flow passage area, which in turn is approximately proportional to solidity if inner and outer walls are neglected, this type of viscous loss would be greatest for the subject design. It is shown in reference 3 from compressor-blade cascade data that the cascade total-pressure loss varies approximately in direct proportion with solidity. In an attempt to eliminate the effect of varying solidity and to obtain a better indication of the effect of diffusion for the subject turbine and the two reference turbines, the design-point performance values are compared in

figure 9 on a basis of specific blade loss L, where L is defined as $(1 - \eta_t)/\sigma_m$. The specific blade loss increases substantially between diffusion parameters of 0.15 and 0.30 (from 0.053 to 0.088) and increases much less between 0 and 0.15 (from 0.047 to 0.053). Although the three points do not uniquely define this curve, it appears that the trend might be similar to that obtained for compressor blading (ref. 3) in that the loss increases gradually until a certain value of diffusion is reached and then increases rapidly thereafter.

As pointed out in the TURBINE DESIGN section, the zero-diffusion turbine had a maximum surface critical velocity ratio of 1.08, whereas the two reference turbines were designed for a maximum value of 1.25. From the comparison of the results obtained with the three turbines (fig. 9), it appears that reducing the design maximum critical velocity ratio resulted in only a second-order effect on performance; and it is felt that the difference in performance, as compared on this basis, resulted mainly from the change in the design diffusion that was employed.

From the correlation curve of figure 9 it can be seen that, for a given diffusion-parameter value, the over-all efficiency η_t could theoretically be changed by varying the solidity within certain limits. Because the solidity varied among the three turbine designs compared herein, the differences noted in their over-all performance are felt to result from the different solidities employed, as well as from the changes in the design diffusion parameter. However, if the design diffusion parameter were varied over the same range for a constant solidity, the effect of diffusion on over-all performance would be expected to be even greater, since for the turbines compared herein the solidity was decreased as the diffusion parameter was increased. Thus it is indicated that the diffusion parameter is an important design consideration that affects the over-all turbine performance.

SUMMARY OF RESULTS

A transonic turbine which was designed for zero diffusion of the suction-surface velocity has been investigated experimentally and the pertinent results are as follows:

1. At design equivalent work and design speed, the turbine total-pressure-ratio efficiency was between 0.86 and 0.87 and the highest efficiency was 0.87. The rating efficiency at the design point was between 0.85 and 0.86.

2. The over-all performance of the subject turbine was compared with that obtained with two other transonic turbines having diffusion parameters of 0.15 and 0.30. This comparison showed that the performance of

the zero-diffusion turbine represents an improvement of about $1\frac{1}{2}$ efficiency points over that of the turbine having a design diffusion parameter of 0.15 and about $5\frac{1}{2}$ points over that of the turbine having a diffusion parameter of 0.30. Although this increase in efficiency was not uniquely caused by the decrease in the diffusion parameter since other design variables were changed among the three turbines, it is indicated that the diffusion parameter is an important design consideration that affects the over-all turbine performance.

3. A comparison was also made on the basis of specific blade loss of the design-point performance of the subject turbine with that of the two transonic turbines previously investigated. From this comparison it was indicated that reducing the diffusion parameter from 0.15 to zero resulted in a decrease in specific blade loss from 0.053 to 0.047, whereas the reduction in diffusion parameter from 0.30 to 0.15 affected a more substantial reduction in specific blade loss from 0.088 to 0.053.

4. From the design-point-survey results it was shown that the drop-off in efficiency level in the outer portion of the blade height was not as pronounced for the subject turbine as that noted for the other two transonic turbines in previous investigations. This effect was felt to be mainly attributable to lower diffusion employed in the design of the subject turbine.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, June 16, 1954

REFERENCES

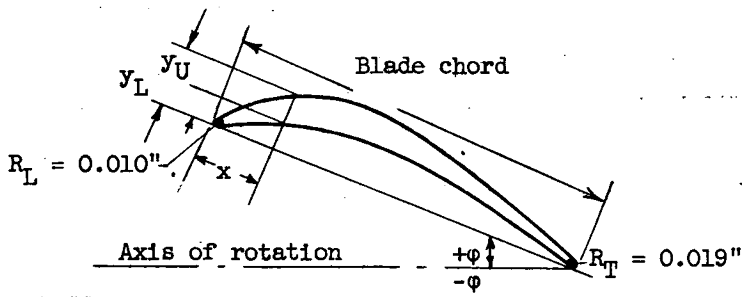
1. Stewart, Warner L., Wong, Robert Y., and Evans, David G.: Design and Experimental Investigation of Transonic Turbine with Slight Negative Reaction Across Rotor Hub. NACA RM E53L29a, 1954.

2. Wong, Robert Y., Monroe, Daniel E., and Wintucky, William T.: Investigation of Effect of Increased Diffusion of Rotor-Blade Suction-Surface Velocity on Performance of Transonic Turbine. NACA RM E54F03, 1954.

3. Lieblein, Seymour, Schwenk, Francis C., and Broderick, Robert L.: Diffusion Factor for Estimating Losses and Limiting Blade Loadings in Axial-Flow-Compressor Blade Elements. NACA RM E53D01, 1953.

4. Whitney, Warren J., Buckner, Howard A., Jr., and Monroe, Daniel E.: Effect of Nozzle Secondary Flows on Turbine Performance as Indicated by Exit Surveys of a Rotor. NACA RM E54B03, 1954.

TABLE I. - ROTOR-BLADE-SECTION COORDINATES



	Hub		Mean		Tip	
φ, deg	-3.25		7.93		19.87	
r/r_t	0.70		0.85		1.00	
x, in.	y_U, in.	y_L, in.	y_U, in.	y_L, in.	y_U, in.	y_L, in.
0	0.010	0.010	0.010	0.010	0.010	0.010
.100	.142	.088	.138	.081	.116	.061
.200	.257	.182	.255	.167	.213	.123
.300	.365	.264	.357	.240	.297	.176
.400	.465	.337	.449	.304	.371	.223
.500	.555	.400	.534	.359	.435	.264
.600	.637	.455	.606	.405	.487	.299
.700	.710	.504	.666	.443	.531	.330
.800	.770	.546	.715	.474	.566	.357
.900	.819	.581	.752	.498	.593	.379
1.000	.857	.609	.777	.518	.612	.397
1.100	.884	.632	.792	.533	.623	.412
1.200	.901	.651	.798	.543	.630	.423
1.300	.908	.666	.795	.547	.631	.430
1.400	.904	.675	.784	.547	.625	.433
1.500	.892	.676	.765	.543	.613	.430
1.600	.870	.669	.738	.533	.595	.423
1.700	.839	.653	.703	.517	.571	.412
1.800	.800	.630	.663	.496	.541	.396
1.900	.753	.599	.617	.469	.507	.376
2.000	.696	.560	.565	.438	.470	.353
2.100	.632	.513	.510	.403	.431	.327
2.200	.561	.459	.452	.362	.389	.298
2.300	.486	.398	.391	.316	.346	.267
2.400	.406	.330	.329	.266	.303	.234
2.500	.325	.258	.267	.213	.260	.199
2.600	.242	.182	.205	.156	.217	.163
2.700	.159	.104	.143	.096	.174	.126
2.800	.076	.025	.081	.035	.131	.087
2.858	.019	.019	----	----	----	----
2.885	----	----	.019	.019	----	----
2.900	----	----	----	----	.088	.046
3.000	----	----	----	----	.045	.004
3.032	----	----	----	----	.019	.019

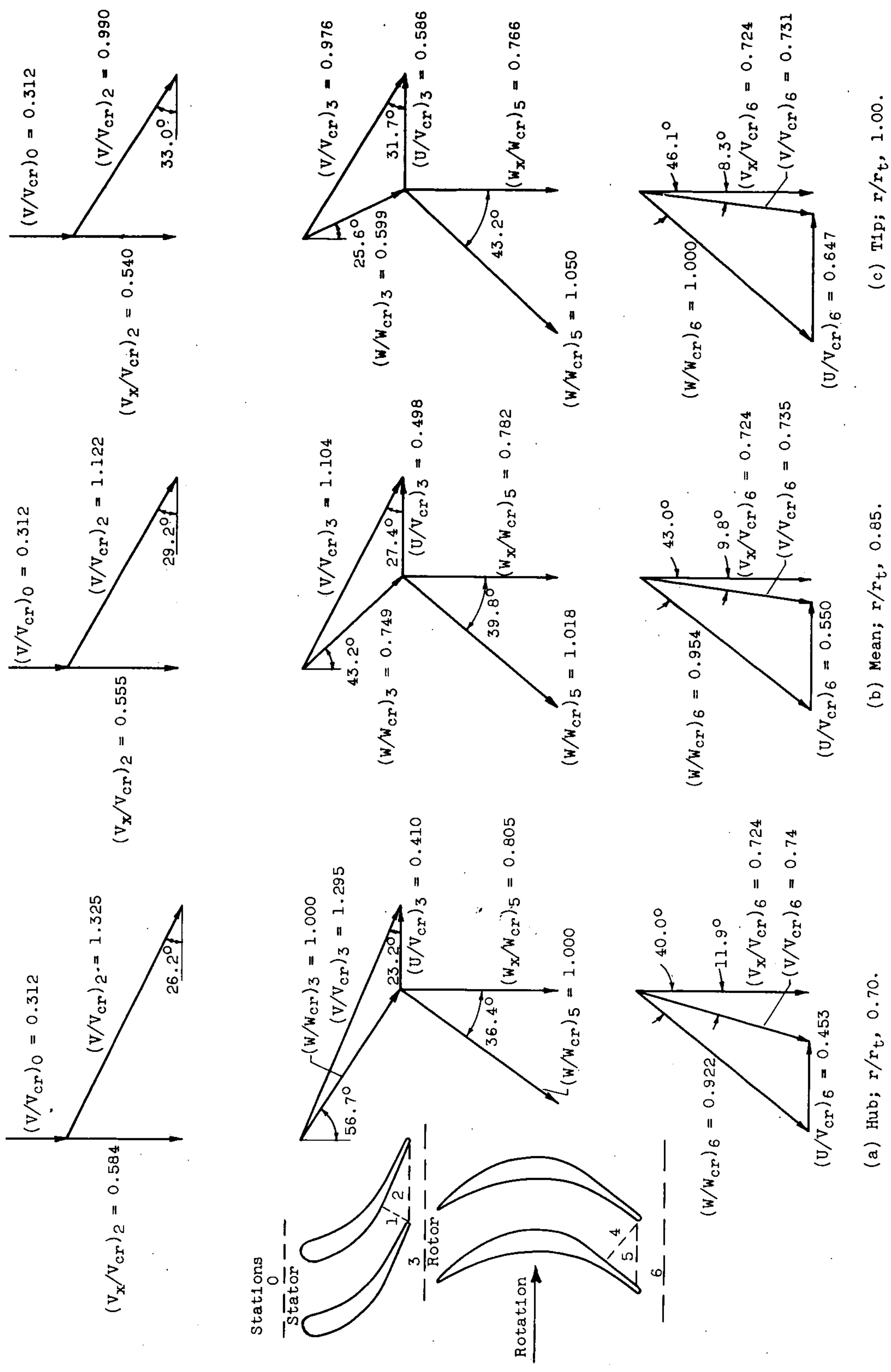


Figure 1. - Transonic-turbine velocity diagrams.

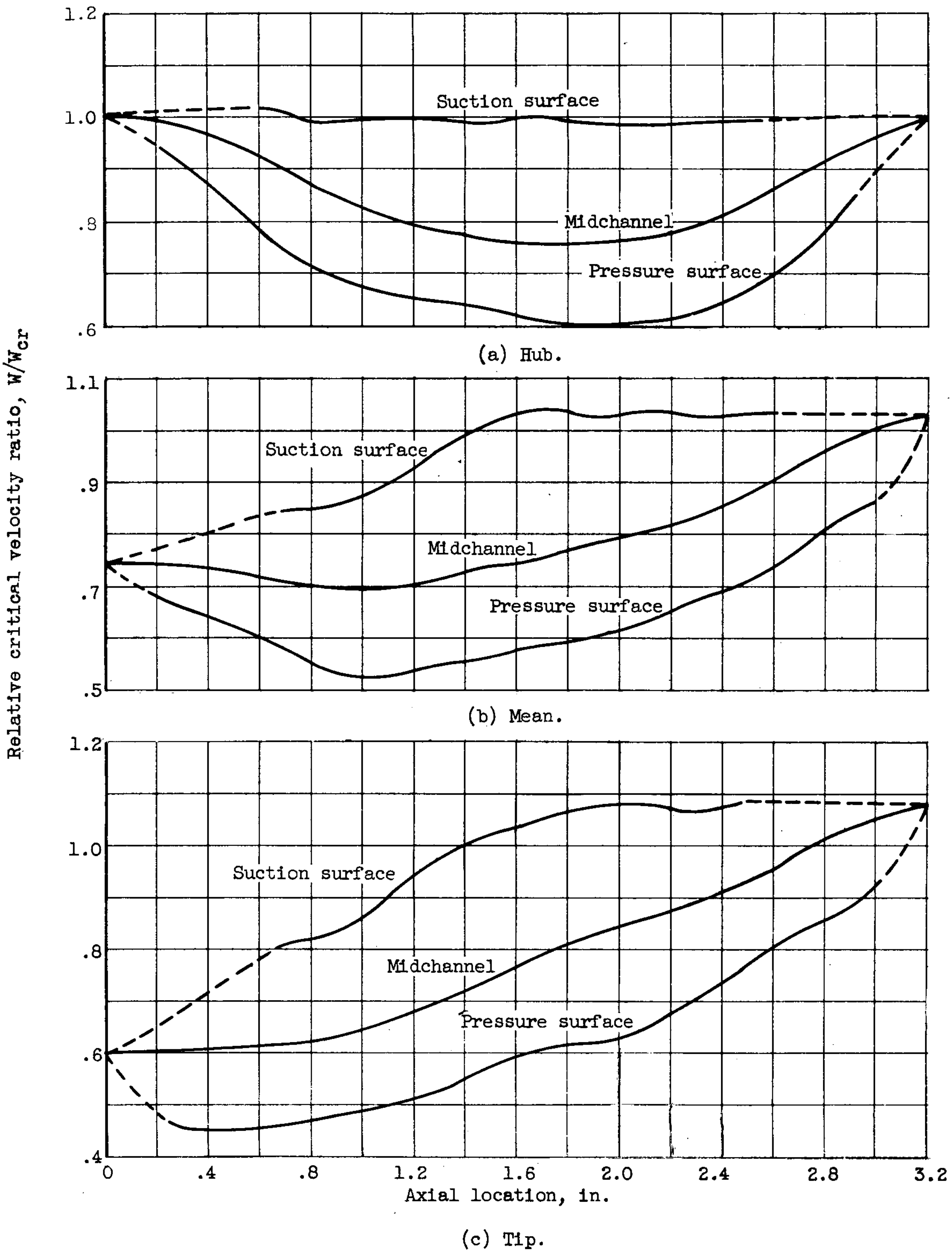
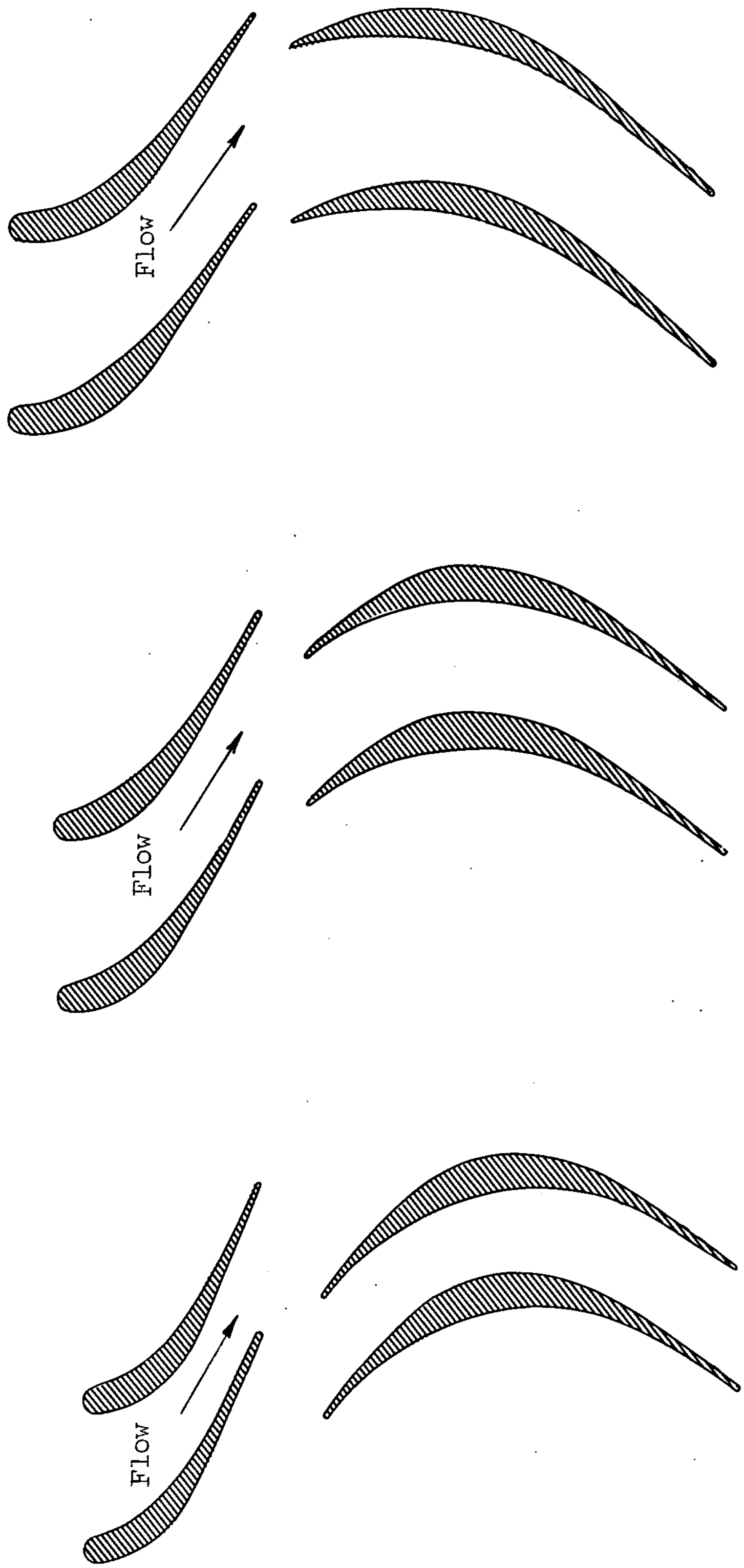


Figure 2. - Design rotor midchannel and surface velocity distributions at hub, mean, and tip sections. (Dashed line denotes extrapolation to blade inlet and blade outlet values.)



(a) Hub section.

(b) Mean section.

(c) Tip section.

Figure 3. - Stator- and rotor-blade profiles and flow passages.

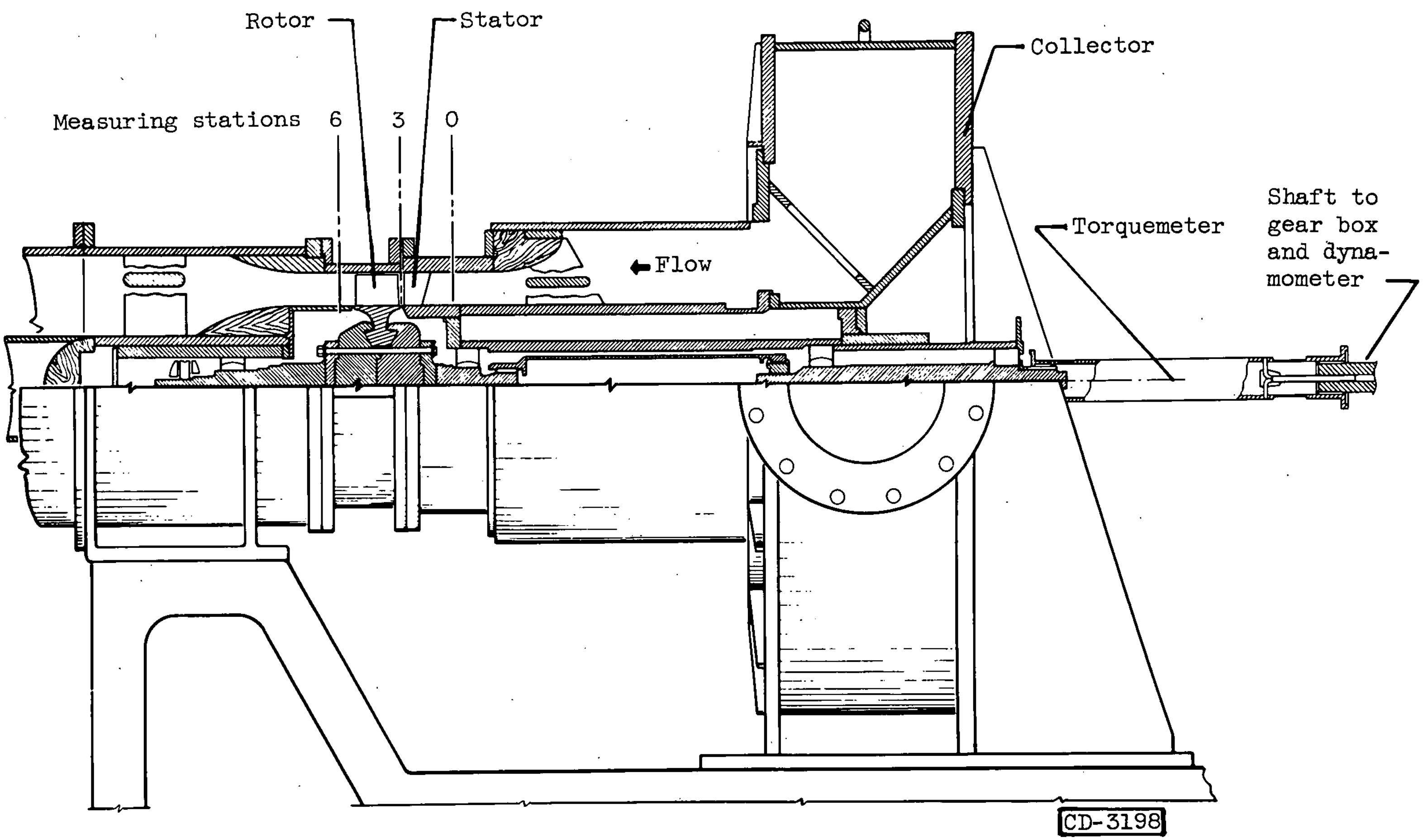


Figure 4. - Diagrammatic sketch of cold-air turbine test section.

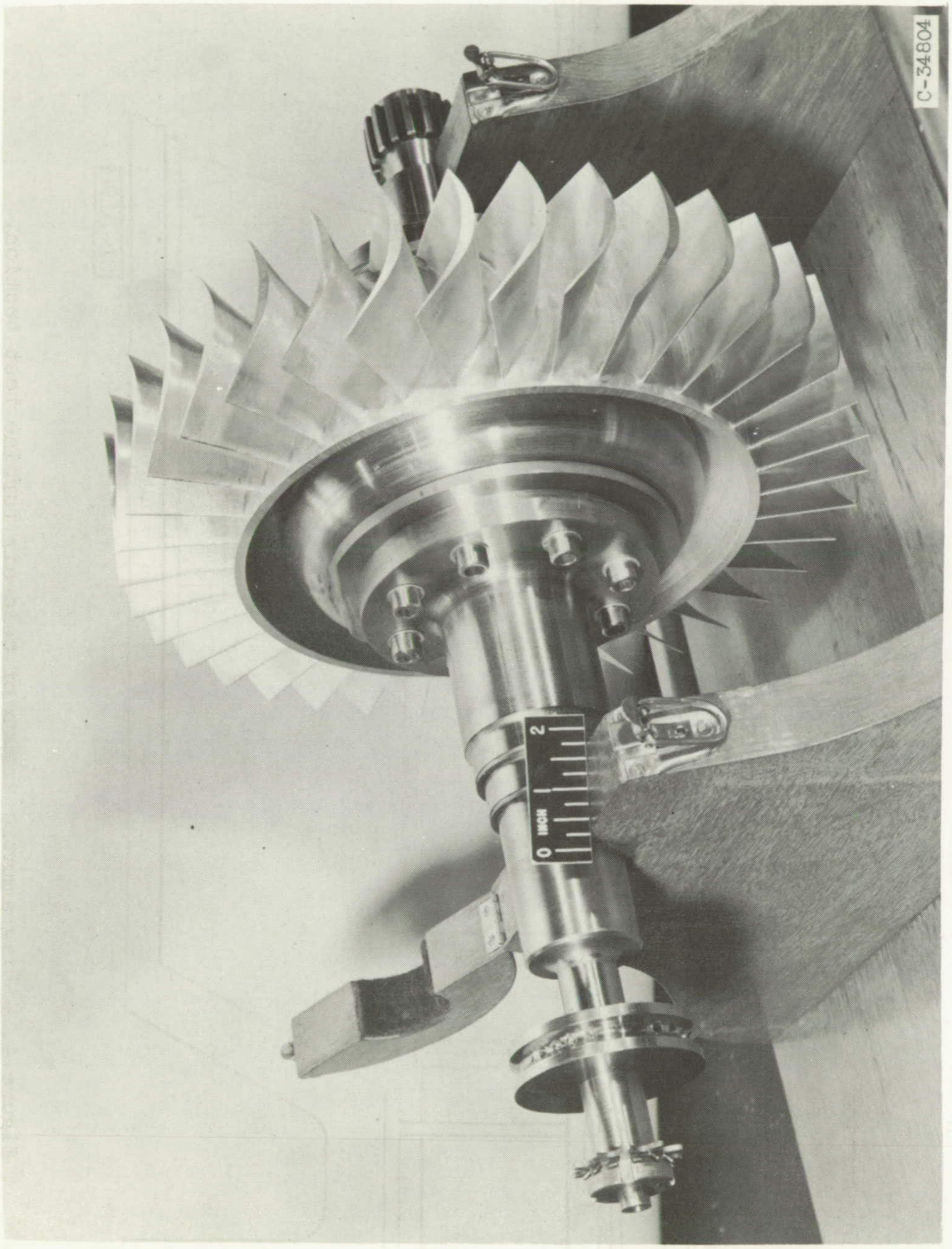
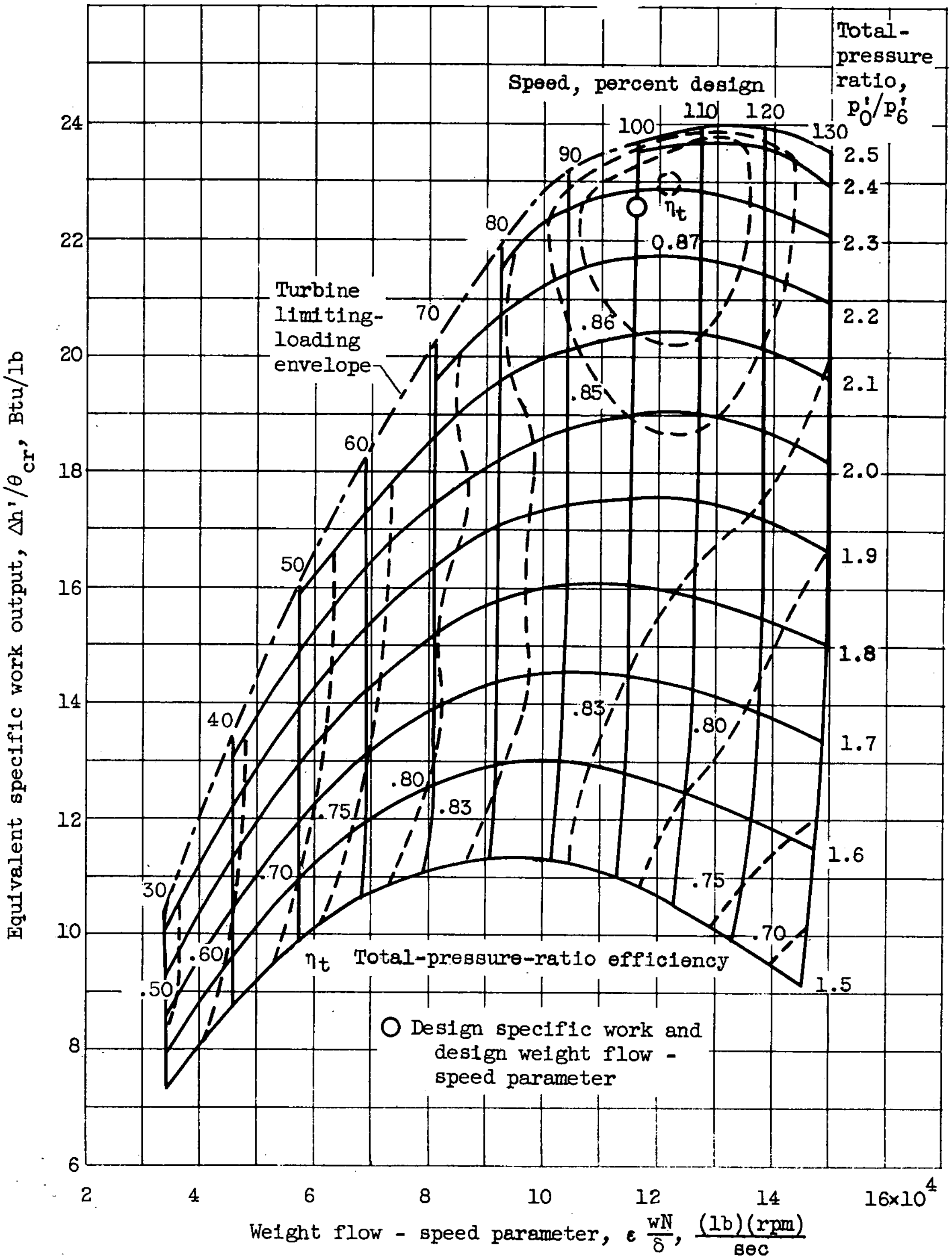
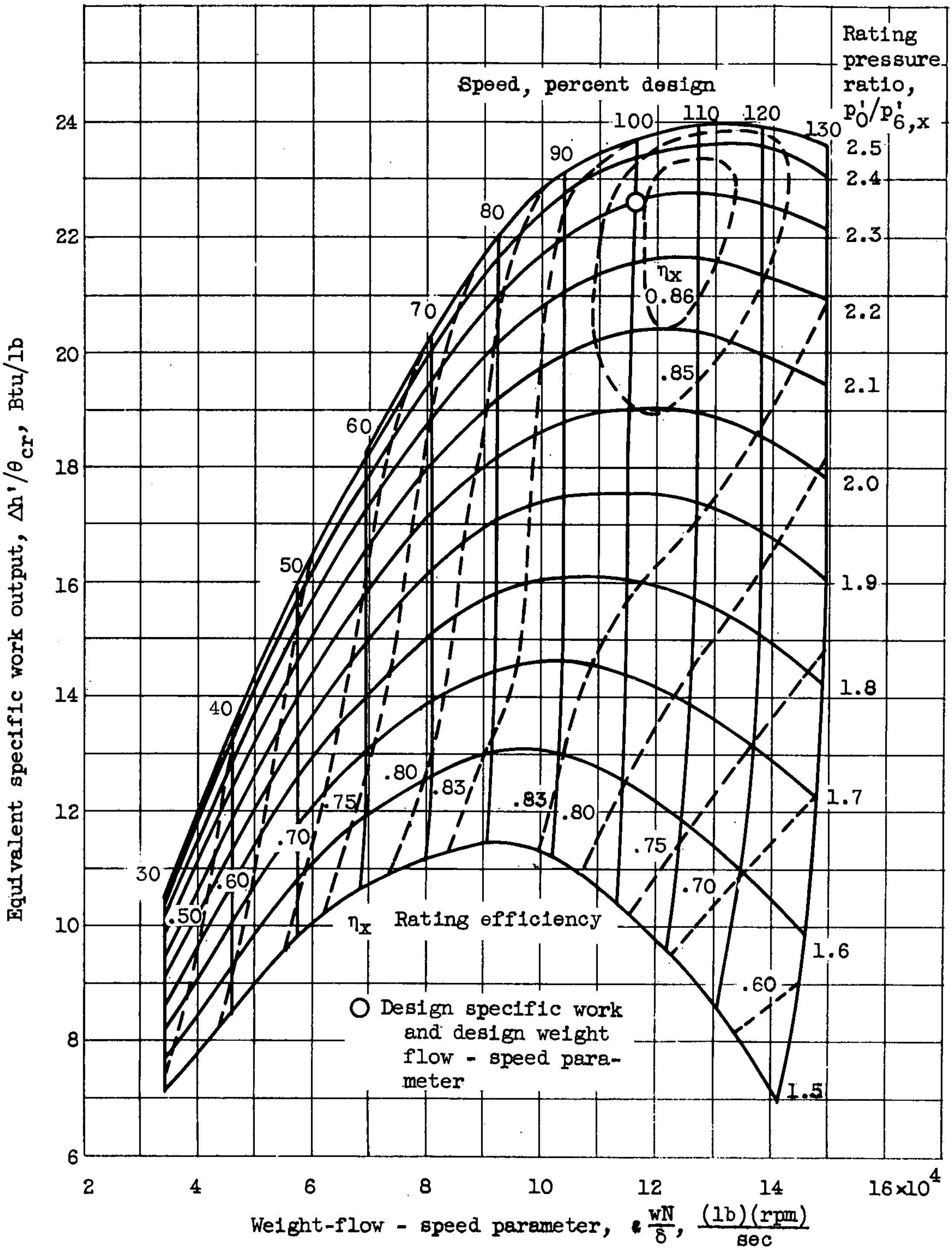


Figure 5. - Photograph of transonic-turbine rotor assembly



(a) Based on total-pressure ratio.

Figure 6. - Over-all turbine performance.



(b) Based on rating total-pressure ratio.

Figure 6. - Concluded. Over-all turbine performance.

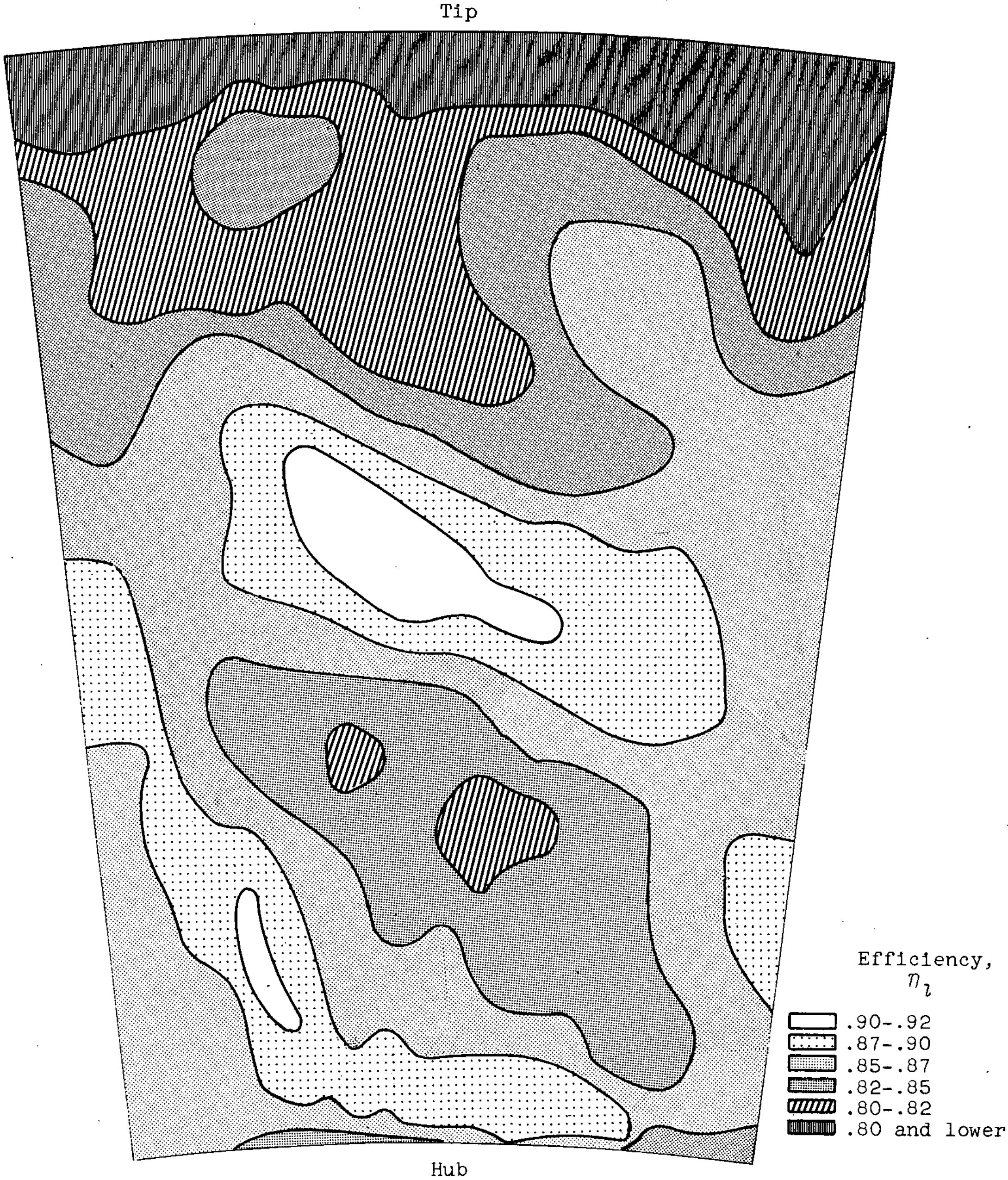


Figure 7. - Contours of local adiabatic efficiency η_l across turbine from detail surveys taken at design operating conditions. (Efficiencies shown for portion of turbine-outlet flow annulus corresponding to $1\frac{1}{4}$ stator passages.)

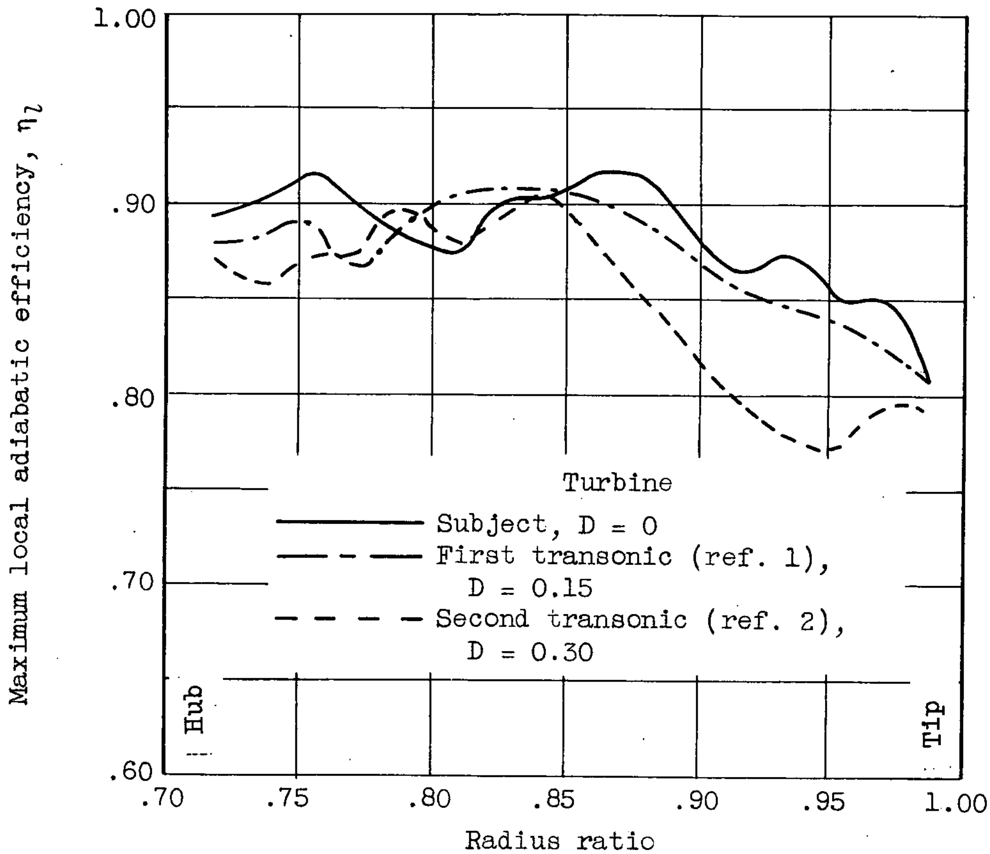


Figure 8. - Variation of maximum local adiabatic efficiency with radius ratio for three transonic-turbine configurations.

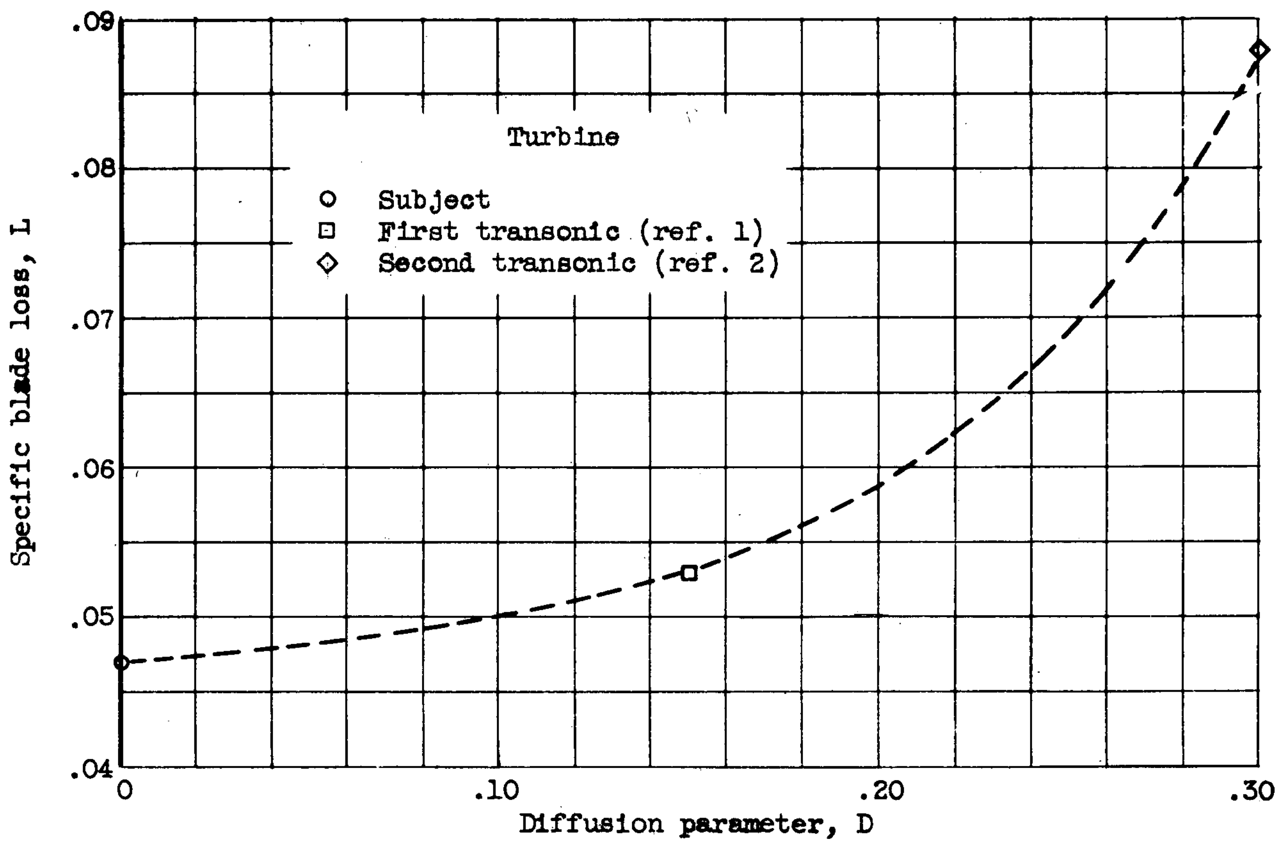


Figure 9. - Effect of diffusion parameter on specific blade loss as determined from design-point performance for three transonic-turbine configurations.